1990

Some Thermodynamic Performance Test Results of Refrigerant 134a

W. K. Snelson
National Research Council Canada

J. W. Linton
National Research Council Canada

P. F. Hearty
National Research Council Canada

T. N. Duong
University of Sherbrooke

SOME THERMODYNAMIC PERFORMANCE TEST RESULTS OF REFRIGERANT 134a

W.K. Snelson P.Eng., J.W. Linton P.Eng., P.F. Hearty

National Research Council Canada
Low Temperature Laboratory
Ottawa, Ontario, K1A 0R6

T.N. Duong

University of Sherbrooke
Sherbrooke, Quebec, J1K 2R1

ABSTRACT

As part of an ongoing investigation into environmentally acceptable alternatives to the CFC refrigerants a series of tests have been completed comparing the thermodynamic performance of R134a with R12. The tests were carried out under controlled conditions in a well instrumented water/water heat pump test facility, using an open-drive reciprocating compressor and counter flow heat exchangers.

Based on the compressor design and chosen cycle conditions theoretical predictions of evaporator capacities for these two fluids indicate that a crossover point should occur at a particular evaporating temperature, with the corresponding capacity of R134a being less than R12 below that point and greater than R12 above the crossover. This prediction has been verified from the experimental test results. The test data are also used to compare other important performance characteristics of R134a and R12 including coefficients of performance (COPs), refrigerant mass flows, and volumetric refrigeration effects. The paper also includes a brief description of the test facility and the measurement techniques used.

The test results are representative of various refrigeration cycle conditions and cover an evaporating temperature range from -4°F (-20°C) to 50°F (10°C), with a constant condensing temperature of 122°F (50°C). The degree of subcooling and the superheat at compressor inlet were maintained constant throughout the tests.

INTRODUCTION

Some preliminary test measurements conducted at this laboratory and reported previously (Linton and Snelson 1989) indicated that lower overall cycle efficiencies should be expected for refrigerant R134a when considered as a drop-in substitute for R12. For convenience purposes those tests were carried out in a small prototype exhaust air heat pump (EAHP) whose charge requirements were consistent with the small sample quantity of R134a available at that time. The design characteristics of the EAHP (Linton 1986) are such that the refrigerant evaporating temperature remained approximately constant over the range of ambient outdoor air temperatures used. Thus the early EAHP performance comparison tests were applicable over a very narrow band of evaporating temperatures around 40°F (4.4°C).

In the current series of tests reported here a highly instrumented water/water heat pump test facility was used to provide a more detailed comparison of the thermodynamic performances of R12 and R134a. The flexibility of this equipment enabled the investigation to cover a broad range of evaporating temperatures representative of the normal span of conditions anticipated in refrigeration and air-conditioning system applications.

Theoretical performance comparisons conducted by others (Wilson and Basu 1988) have indicated that a crossover condition is to be expected in evaporating capacity performances of these fluids. For evaporating temperatures below the crossover point it is predicted that R134a will have a lower capacity than R12, but will have a higher capacity above the crossover. In this paper the corresponding theoretical crossover point is calculated for a typical set of reference cycle conditions used throughout the tests. The experimental test results are used to verify this prediction and to compare key performance characteristics for the two refrigerants.

This series of tests was undertaken as part of an ongoing joint collaboration arrangement between the National Research Council Canada, DuPont Canada Inc., and Energy, Mines and Resource Canada.

EXPERIMENTAL EQUIPMENT

Test Facility

Figure 1 shows a pictorial view of the Water/Water Heat Pump Test Facility. A schematic flow diagram of the facility is shown in Figure 2. It consists basically of three closed loops.

The refrigerant circuit uses an open-drive two cylinder reciprocating compressor driven by a variable speed 5 hp (3730 W) electric motor. An accumulator and oil separator are provided on the compressor suction and discharge lines respectively. Refrigerant flow is controlled manually using an electrically operated expansion valve. The heat exchangers are counter flow tube-in-tube configurations and are each divided into four horizontal straight equal sections of 40 in (1016 mm) long copper tubing, joined together with short U-tube interconnecting pieces. In the evaporator the refrigerant flows inside double fluted tubes of 1 1/8 in (28.6 mm) OD, which are located inside smooth tubes of 1 1/4 in (31.7 mm) ID. The refrigerant flows through the surrounding annuli in the condenser sections. All heat exchanger sections are surrounded by 6 in (152 mm) of fibreglass insulation. Subdividing of the heat exchangers into sections, with instrumentation at each inlet and outlet, enables the facility also to be used to study the evaporation and condensation behaviour of mixtures of refrigerants in a heat pump system.

The heat source is provided to the evaporator by a water-glycol circuit which contains a storage tank, circulating pump, flowmeter, electric heater, and interchanger. The temperature of the water-glycol system is controlled by the heater, and flow to the evaporator can be modulated using throttle valves and a bypass line. Water can be used in place of water-glycol for tests using higher refrigerant evaporating temperatures.

Heat is transferred from the condenser into a water circulation loop, in which water is pumped from a storage tank through the inside tubes of the condensing sections. The water loop also contains a flowmeter and interchanger, and the water temperature is controlled by partial heat rejection to a chilled water system in a separate heat exchanger. The interchanger is used to transfer most of the heat produced in the condenser to heat the water-glycol circuit back up again. Water flow rates into the condenser can also be regulated by valves and a bypass line.

Instrumentation

Platinum RTD precision Digital Thermometers with an accuracy of ±0.018°F (±0.01°C) are used to measure temperatures of water and water-glycol entering and leaving the various heat exchanger sections, where the differences between inlet and outlet temperatures are likely to be small. Key instream refrigerant temperatures are measured with copper-constantan thermocouples inserted into thermowells positioned in the bulk of the refrigerant flows. The uncertainty of the thermocouple temperature measurements is ±0.9°F (±0.5°C). Elsewhere system temperatures are measured with thermocouples soldered directly to the outside of the copper tubing and covered with at least 1 in (25.4 mm) of foam insulation.

Figure 1. Water/Water Heat Pump Test Facility

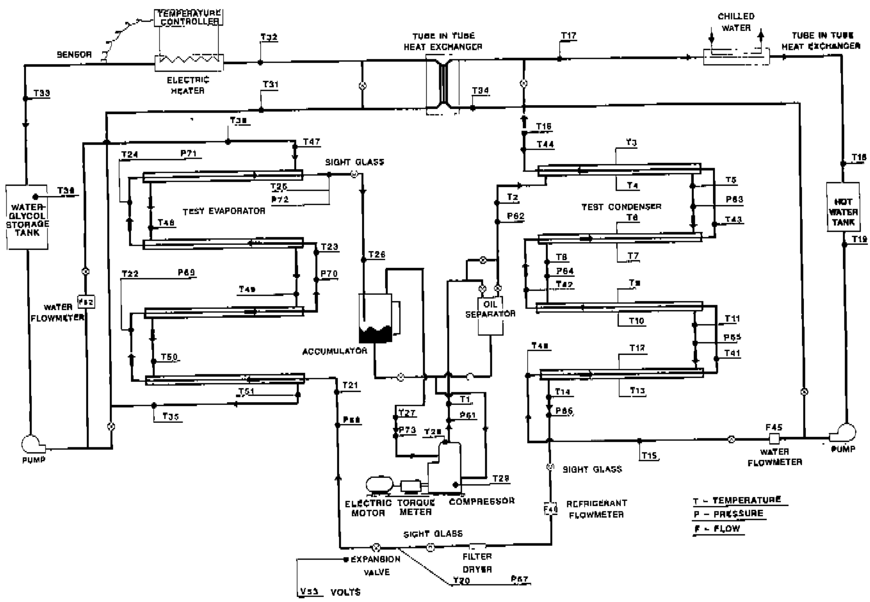

Figure 2. Schematic Flow Diagram of the Water/Water Heat Pump Test Facility

Refrigerant pressures are measured using pressure transducers connected to static pressure taps located at the inlet and outlet of each heat exchanger section. The pressure transducers are calibrated to ±1.45 psi (±10 kPa).

Refrigerant mass flow is measured directly with a Coriolis effect mass flow meter mounted in the liquid line leaving the condenser. This meter is calibrated by the manufacturer and provides an accuracy of ±0.044 lb/min (±0.02 kg/min). A similar type of meter is used to measure the flow rate in the water-glycol circuit used on the evaporator side of the test loop. The water flow rate in the condenser side loop is measured with a paddle wheel type flow meter, calibrated in place using the bucket and stop-watch method. The estimated accuracy of this device is ±0.22 lb/min (±0.1 kg/min).

Power input to the compressor is measured with a torque sensor mounted between the electric motor and compressor drive shaft. The sensor is a strain gauge type and its accuracy is 0.1% of full scale, ie. ±0.5 lb in ($\pm 5.6 \times 10^{-2}$ Nm) in this case.

Data Acquisition

All data inputs and test parameters are measured by a Hewlett-Packard model 3497A Data Acquisition/Control Unit and processed by a Hewlett-Packard model 9845B Desktop Computer. The only exceptions are the precision Digital Thermometers which communicate directly with the 9845B computer via an IEEE-488 interface bus.

TEST MEASUREMENTS

The refrigerant condensing temperature was maintained constant at 122°F (50°C) throughout the tests by controlling the flow and temperature of water into the condensing section. The degree of subcooling at the condenser outlet was also held constant at approximately 20°F (11.1°C) by adjusting the total refrigerant charge in the system.

The refrigerant evaporating temperatures were set over a range of -10°F to 50°F (−23.3°C to 10°C) by adjusting the temperature and flow rate of water-glycol entering the evaporator. Saturated vapour conditions were maintained at the evaporator outlet for each test run. These condition were established by observation of evaporator outlet line refrigerant temperatures, and subsequent gradual adjustment of the electric expansion valve. Pressure drop in the evaporator outlet lines and accumulator section together with heat gained from ambient room conditions served to provide a superheat of approximately 7°F (3.9°C) at compressor inlet. The same level of superheat was maintained throughout the tests. The compressor speed was held constant at 1950 RPM.

All test readings were taken under steady state conditions which were reached typically in about one hour. The data acquisition system scanned all channel inputs frequently while the system was coming to steady state. When a satisfactory condition was reached the corresponding scan data was retrieved, and the data acquisition program used to process the raw data and make the necessary system performance calculations.

On completion of the baseline tests with R12, the compressor and system were drained, flushed with R11, and evacuated several times using first nitrogen and then R134a to break the vacuum, all in accordance with a recommended procedure supplied by the manufacturer of the new refrigerant. The system was then recharged with R134a, and the alkylbenzene-based lubricating oil was replaced by a polyalkyleneglycol synthetic oil with a viscosity of 150 SUS (32 mm^2/s) at 104°F (40°C) supplied for this purpose by the refrigerant manufacturer. The same series of tests were then repeated for R134a.

PERFORMANCE PREDICTION

Refrigerating Capacity

The refrigerating capacity Q of the system is defined as

$$Q = m \times \Delta h \tag{1}$$

where m = refrigerant mass flow rate
Δh = refrigerating effect

The system mass flow rate delivered by the compressor is given by

$$m = \rho \times V \times \eta_v \times N \times n$$

where ρ = suction vapour density at compressor inlet
V = piston swept volume
η_v = volumetric efficiency of compressor
N = crankshaft revolutions per unit time
n = number of cylinders

For the same compressor, V, N, and n are equal for R12 and R134a.

Therefore the mass flow ratio for the two fluids is given by

$$\frac{m_{R12}}{m_{R134a}} = \frac{\rho_{R12}}{\rho_{R134a}} \times \frac{\eta_{vR12}}{\eta_{vR134a}} \tag{2}$$

For a basic vapour compression cycle (ASHRAE 1989)

$$\eta_v = 1 + C\left(1 - \frac{v_s}{v_d}\right)$$

where C = clearance ratio
v_s = suction specific volume of refrigerant
v_d = discharge specific volume of refrigerant

Equation (2) becomes

$$\frac{m_{R12}}{m_{R134a}} = \frac{\rho_{R12}}{\rho_{R134a}} \times \frac{1 + C\left(1 - (\frac{v_s}{v_d})_{R12}\right)}{1 + C\left(1 - (\frac{v_s}{v_d})_{R134a}\right)} \tag{3}$$

Referring to Figure 3

$$\Delta h = h_4 - h_2 = h_4 - h_1 \tag{4}$$

For a given evaporating temperature and suction superheat, and ignoring pressure losses in the suction line, corresponding values of h_4, υ_s, and ρ can be determined for each refrigerant.

For constant condensing temperature and liquid subcooling conditions corresponding values of h_1 can be established.

Conditions at compressor discharge can be predicted by assuming a relationship for isentropic efficiency η_{is} (Duminil 1976), ie.

$$\eta_{is} = 1 - 0.05 \times \frac{P_d}{P_s}$$

where P_d = discharge pressure
P_s = suction pressure

Assuming suction and discharge pressures are approximately equal to evaporating and condensing pressures respectively, η_{is} can be calculated for any set evaporating temperature.

Enthalpy h_5 at compressor outlet can be calculated using η_{is}.

Knowing h_5 and P_d, the corresponding vapour specific volume υ_d can be determined by interpolation from the refrigerant properties.

Defining the refrigerating capacity ratio of R12 to R134a as Ψ

from (1) $$\Psi = \frac{Q_{R12}}{Q_{R134a}} = \frac{m_{R12}}{m_{R134a}} \times \frac{\Delta h_{R12}}{\Delta h_{R134a}}$$

Mass flow ratios are determined from equation (3) and values of Δh from equation (4). Thus for any given evaporating temperature the corresponding value of Ψ can be calculated knowing the relevant properties of the fluids, and assuming a constant condensing temperature of 122°F (50°C), constant liquid subcooling of 20°F (11.1°C), and constant suction superheat of 7°F (3.9°C). For R12 the required fluid properties were taken from (Downing 1965), and for R134a the data of (Wilson and Basu 1988) were used. Clearance ratio for the compressor was taken to be 2%.

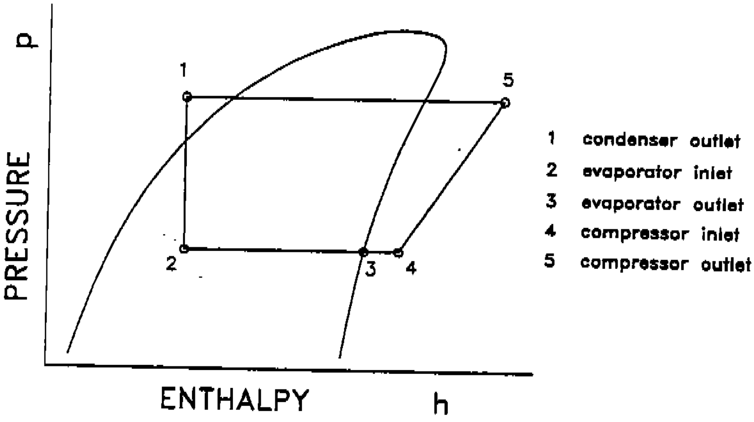

Figure 3. Pressure - Enthalpy Diagram for Simplified Refrigeration Cycle

Table 1 shows the corresponding values of Ψ and other derived parameters calculated for the selected range of evaporating temperatures T_e, and it can be seen that Ψ varies between 0.963 and 1.112. When $\Psi = 1$ the refrigerating capacities Q_{R12} and Q_{R134a} are equal, and this corresponds to the crossover point. In this case it is predicted that a crossover condition should occur at an evaporating temperature of 33.7°F (1°C).

T_e °F	ρ_{R12} lbs/ft³	ρ_{R134a} lbs/ft³	η_{vR12}	η_{vR134a}	Δh_{R12} Btu/lb	Δh_{R134a} Btu/lb	Ψ
-5.0	0.55	0.41	0.91	0.89	45.63	56.79	1.112
0.0	0.61	0.46	0.92	0.90	46.17	57.52	1.096
5.0	0.67	0.51	0.92	0.90	46.70	58.24	1.080
10.0	0.74	0.57	0.93	0.91	47.23	58.96	1.065
15.0	0.81	0.63	0.94	0.92	47.76	59.67	1.050
20.0	0.89	0.70	0.94	0.93	48.28	60.38	1.036
25.0	0.98	0.77	0.95	0.94	48.80	61.09	1.022
30.0	1.07	0.85	0.95	0.94	49.32	61.78	1.009
35.0	1.16	0.94	0.96	0.95	49.83	62.48	0.997
40.0	1.27	1.03	0.96	0.95	50.34	63.16	0.985
45.0	1.38	1.13	0.97	0.96	50.84	63.84	0.974
50.0	1.50	1.24	0.97	0.96	51.33	64.51	0.963

Table 1. Predicted Refrigerating Capacity Ratios (Ψ) of R12 to R134a for Fixed Condensing Temperature of 122°F (50°C)

TEST RESULTS AND DISCUSSION

The experimental values of evaporator capacities were obtained from mass flow measurements in the water-glycol circuit, and from accurate temperature difference measurements of the flow between inlet and outlet. Precise knowledge of water-glycol specific heat as a function of temperature was also necessary. The corresponding evaporator capacities were calculated for each evaporating temperature used in the series of tests, for both refrigerants. These results were plotted in Figure 4 and it can be observed that a crossover situation does occur for the two curves at an evaporating temperature of approximately 31°F (−0.55°C). This crossover point in performance for R12 and R134a shows close agreement with the predicted condition obtained from Table 1. This indicates that the assumptions made and approximate relationships used for calculation of compressor efficiencies were valid. The crossover condition is a result of the combined effect of the properties of enthalpy and vapour density existing in the compressor suction inlet.

Compressor shaft power inputs were determined from torque meter readings obtained under steady state conditions during the tests. Plotting of these results (Figure 5) also indicates that a crossover condition occurs for the two refrigerants, in this case at about 12°F (−11.1°C). The ratio of power inputs for the two fluids is also determined by corresponding properties of enthalpy and density.

Combining the results of Figures 4 and 5 produces the corresponding Coefficient of Performance (COP) for cooling (Figure 6). The net effect shows no crossover in this case, but indicates a consistently lower COP in the case of R134a, with perhaps a tendency to converge at higher evaporating temperatures.

Heating capacities were derived from measurements of water flow and temperature increase in the condenser loop. Plotting of these results in Figure 7 clearly shows that a crossover condition exists again, this time at an evaporating temperature around 25°F (−3.9°C). Corresponding COP-Heating results derived from Figures 5 and 7 and plotted in Figure 8 indicate the same trend as that for COP-Cooling revealed in Figure 6.

Measurements of refrigerant mass flow rates of R12 and R134a were plotted in Figure 9. The lower values are to be expected in the case of R134a due to its much lower suction vapour density and higher values of heat of vapourization.

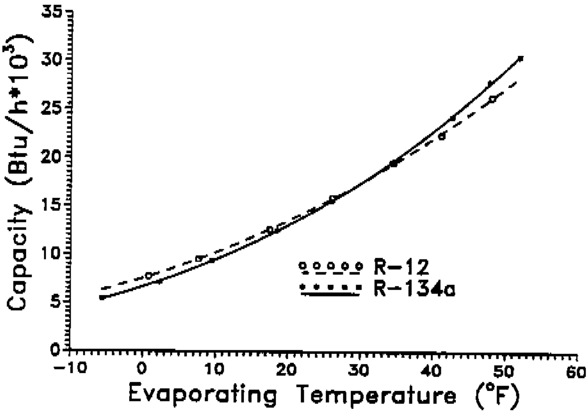

Fig. 4 EVAPORATOR CAPACITY

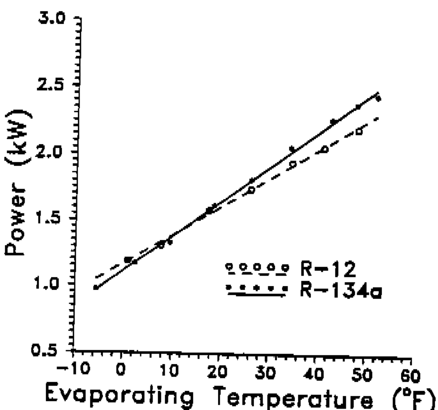

Fig. 5 COMPRESSOR POWER

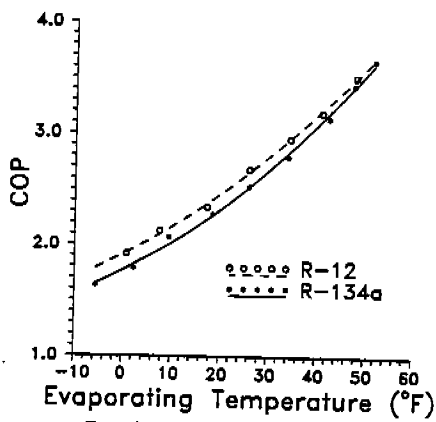

Fig. 6 COP–COOLING

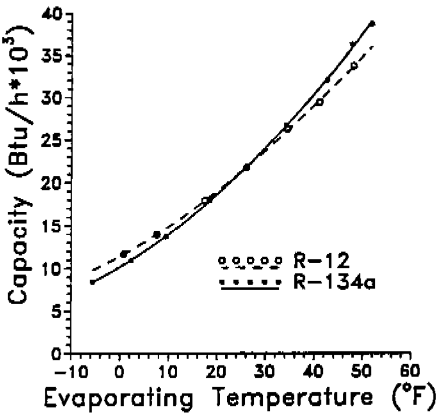

Fig. 7 CONDENSER CAPACITY

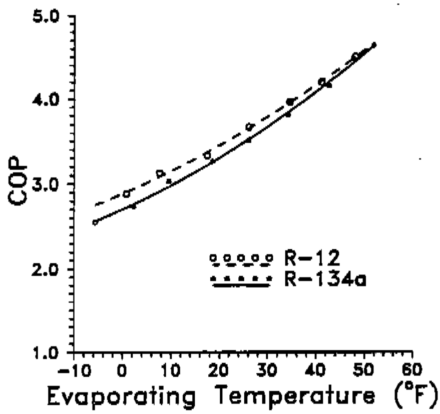

Fig. 8 COP-HEATING

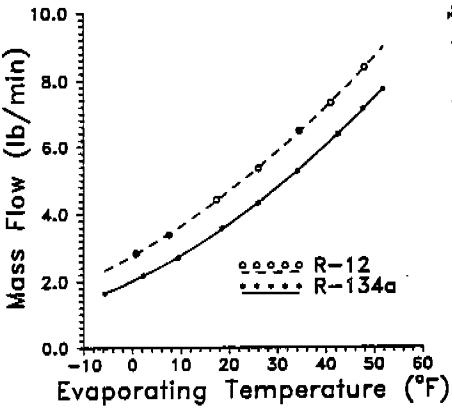

Fig. 9 REFRIGERANT MASS FLOW

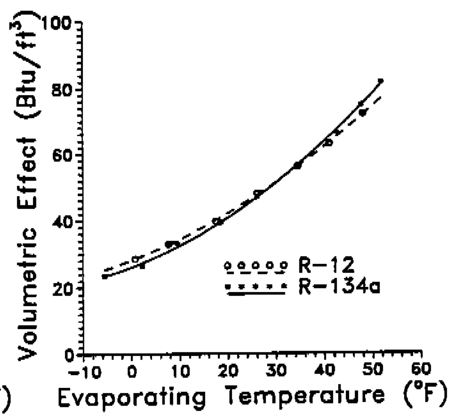

Fig. 10 VOLUMETRIC REFRIG. EFFECT

From the respective refrigerant properties data, the enthalpy and vapour density at compressor inlet can be established from the measurements of suction pressure and temperature obtained for each test condition. Multiplying enthalpy by suction vapour density gives the corresponding volumetric refrigeration effect, ie. the amount of heat removed per unit volume of vapour drawn into the compressor. These results were plotted in Figure 10, which indicates that a crossover condition occurs as in Figure 4 at an evaporating temperature around 31°F (-0.55°C). Since the volumetric efficiency is approximately the same for R12 and R134a at any given evaporating temperature (Table 1), this means that the actual volumes of suction vapour drawn into the compressor per unit time will be the same for both refrigerants at those conditions. Thus it can be seen that the capacities measured and plotted in Figure 4 are proportional to the corresponding volumetric refrigeration effects observed in Figure 10, and crossover therefore occurs at the same point.

CONCLUSIONS

Based on thermophysical properties it was predicted that the capacities of R12 and R134a, as obtained in typical refrigeration cycles, are equal at a particular evaporating temperature.

For evaporating temperatures below this crossover point, R134a has a lower capacity than R12, and a higher capacity above it. In the cycle considered, using a constant condensing temperature of 122°F (50°C), 7°F (3.9°C) suction superheat, and 20°F (11.1°C) condenser subcooling, it was predicted that a crossover condition would occur at an evaporating temperature of 33.7°F (0.9°C). Measured values of capacities for a typical range of evaporating temperatures indicated a crossover for the two fluids occurring at 31°F (–0.55°C).

The test measurements showed capacities of R134a becoming gradually less than R12 as evaporating temperatures get lower, and larger than R12 with higher evaporating temperatures. Thus, for an evaporating temperature of –5°F (–20.5°C) the capacity of R134a is 15-20% less than R12, whereas at 50°F (10°C) the R134a capacity is 10-15% greater than R12.

Measurements of compressor power showed that a similar crossover condition occurred but at a lower evaporating temperature around 12°F (–11.1°C). The net effect produced a COP (Cooling) which was lower for R134a over the whole range of evaporating temperatures. For the lowest evaporating temperatures the difference in COPs approached 10%, whereas at higher temperatures the COPs tended to converge. This implies that for systems operating with the same mechanical equipment, the effect on COP of using R134a in place of R12 would be greater for refrigeration than for air-conditioning cycles.

REFERENCES

ASHRAE Fundamentals Handbook 1989. F1.9

Downing, R.C. 1965. "Refrigerant equations." Paper No. 2313, ASHRAE Journal, p. 158-169

Duminil, M. 1976. "Basic principles of thermodynamics as applied to heat pumps: Thermodynamic cycles in heat pumps." NATO Advanced Study Institute Series No. 15, Noordhoff International Publishing, Leyden, p.113.

Linton, J.W. 1986. "Design, construction, and testing of an exhaust air heat pump for R-2000 houses." National Research Council Canada, NRC No. 27719.

Linton, J.W. and Snelson, W.K. 1989. "Performance comparison of Refrigerants 134a and 12 in a residential exhaust air heat pump." ASHRAE Transactions, Vol. 95, Part 2.

Wilson, D.P. and Basu, R.S. 1988. "Thermodynamic properties of a new stratospherically safe working fluid - Refrigerant 134a." ASHRAE Transactions, Vol. 94, Part 2.